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ABSTRACT

Because future space missions will require heat rejection subsystems having megawatt capacity, the development of lightweight heat rejection techniques is desirable. Closed systems for waste heat rejection in space are radiative and their capacity is proportional to the fourth power of absolute temperature. Reductions in the surface area and therefore, in the mass of a space radiator are possible by increasing the heat rejection temperature above the temperature of the thermal source. One proposed method of increasing the heat rejection temperature uses a heat pump powered by a cyclic heat engine operating at a temperature above the waste heat source. This heat pumping technique reduces the required direct radiator surface area, but it introduces mass penalties associated with its power supply and its heat rejection system. The use of a heat pump augmented heat rejection system will generally be practical if it reduces the total radiator mass requirement over a suitable baseline design. The mass penalties of the heat pump augmented radiator over a baseline (a flat plate radiator) are considered in this analysis.

INTRODUCTION

The thermal management of large-scale space power systems will require heat rejection subsystems that comprise a large fraction of the total system volume and mass. The problem of placing these systems into orbit will be especially acute until the next generation of STS is operational. The only heat transfer mechanism available to a closed system for waste heat rejection into space is radiation. Radiation heat transfer is proportional to the fourth power of absolute temperature. A plot of surface area versus heat rejection temperature with heat load as a parameter for an idealized flat plate black body radiator is given in Figure 1. Even a modest increase in heat rejection temperature yields a considerable reduction in radiator area. A reduction in radiator surface area can potentially result in a substantial lessening of its mass. Reductions in the surface area and therefore, in the mass of a space radiator are thus possible by increasing the heat rejection temperature above the temperature of the thermal source. One proposed method of increasing the heat rejection temperature uses a heat pump powered by a cyclic heat engine driven by a prime power heat source. Elevating a radiator's heat rejection temperature with a heat pump may reduce the required radiator area, but it also introduces mass penalties associated with the heat pump system.

The heat pump augmented radiator may be useful, even when its total mass is greater than that of the reference radiator. In many instances, the volume constraints for the transportion configuration of a radiator to orbit may be more limited than are the mass contraints. Here the heat pump augmented radiator system need not be lighter than the reference radiator because the added mass penalty would be offset by an achievable volume reduction. One application for this system would be in the reduction of the volume of electronic component cooling radiators. Although an analysis would have to be conducted on a case by case basis to determine the actual volume reduction possible, the potential radiator volume reduction could be substantial. This cooling system might also be used to increase the power density achievable in electronic components for a given operating temperature. Mass savings of the heat pump augmented system over the reference flat plate design are desirable, but not mandatory.

A careful evaluation is necessary to determine when a heat pump might be beneficial. The usefulness of a heat pump augmented radiator in the design of a heat rejection radiator depends upon a number of considerations including: the augmented radiator's mass as compared with an appropriate baseline design, the number of additional failure modes introduced by the heat pump system, and the feasibility of development of lightweight heat pumps and heat engines capable of operation in the temperature range of interest.

Several preliminary studies have been conducted evaluating the feasibility of heat pump augmentation. A study by Dexter and Haskin (1984) calculated the performance of a 100-kW_t heat pump augmented radiator operating at temperatures between 370 K and 470 K. The mass penalty associated with the heat pump was considered in great detail, but a treatment of the mass penalty from the heat pump power supply was not included. A more generalized treatment of the same problem was given by Kerrebrock (1986) who determined the reduction in radiative area possible with heat pump augmented radiator systems using thermodynamic analysis. An expression was proposed to locate the optimum ratio of heat pump to power source heat rejection temperature. However, only the area reduction achievable with heat pump augmentation was considered and, as with the Dexter and Haskin study, the mass penalty from with the power supply was not included. Kerrebrock concluded that reductions in radiator mass were possible, given an efficient heat pump and power source.

The proposed heat pump augmented radiator system uses a prime power heat source (for example, high-pressure hydrogen heated in a high-temperature gas-cooled reactor) to drive a heat engine (perhaps a closed-cycle turbogenerator) that powers a heat pump compressor. Such a system could elevate the radiator temperature a few hundred degrees Celsius, which would considerably reduce the required radiative area. Unfortunately, these large temperature changes also result in a drastic reduction in the theoretical efficiency of the heat pump, thereby necessitating more massive machinery. A search of the literature failed to find reference to any heat pump operating at temperatures above the critical point of water. The development of high temperature multi-stage heat pumps, perhaps using liquid metals, such as lithium, or organic fluids, such as toluene, will be necessary prior to application of this concept in high-temperature systems. Because technology development would be needed to make such a concept operational, some clear advantage to this system is needed to justify its development.

FORMULATION OF MODEL

A parametric model is developed to determine the performance of a heat pump augmented radiator under a given set of conditions. A schematic of the proposed radiator system is given in Figure 2. The proposed system transfers heat, Q_1, from T_1, a low-temperature waste heat source, to T_2, the rejection temperature of a primary radiator. Work is assumed to be provided by a cyclic engine operating between T_3 and T_2, using heat from a prime power source. The heat from the prime power source need not be rejected at the same temperature as the heat from the refrigeration system. It is expected that in many cases seperate radiators would be desirable. However, for this analysis one radiator will be assumed to simplify the presentation. The expressions given can be easily modified to accomodate multiple heat rejection temperatures. The power required of an ideal device to transfer a heat load, Q_1, from T_1 to T_2 is expressed as

$$P = Q_1/COP \quad . \tag{1}$$

The heat rejected by the primary radiator must then be

$$Q_2 = Q_1 + P = (1 + 1/COP)\ Q_1 \quad , \tag{2}$$

and because the shaft power produced in a Carnot heat engine is

$$P = Q_3\ \eta_{th} \quad , \tag{3}$$

the total heat rejected required in terms of Q_1 is then

$$Q_T = Q_1\ [1 + 1/COP + 1/(COP\ \eta_{th})] \quad . \tag{4}$$

The performance of a real heat pump is some fraction of the Carnot efficiency. Little study has been conducted on the fractional Carnot efficiency of high-temperature heat pumps. This fraction is to be certainly less than 0.6 and perhaps as low as 0.01 in high-temperature heat pumps. This range of fractional Carnot efficiency will be used as a parameter in this analysis. These actual device efficiencies are expressed as

$$e = \epsilon_r \frac{T_1}{(T_2 - T_1)} \quad , \tag{5a}$$

and

$$\eta = \epsilon_e \frac{(T_3 - T_2)}{T_3} \quad , \tag{5b}$$

for the heat pump and heat engine, respectively. The ratio of the total heat rejected by the heat pump augmented radiator to the heat load from the low-temperature source is then computed from Equations (4), (5a), and (5b)

$$\frac{Q_T}{Q_1} = 1 + \frac{(T_2 - T_1)}{e\, T_1} + \frac{(T_2 - T_1)\, T_3}{(T_3 - T_2)\, T_1\, e\, n} \quad . \tag{6}$$

Heat transfer by radiation from the reference radiator is described by the Stefan-Boltzmann law, expressed as

$$Q_1 = \epsilon\ \sigma\ A\ (T_1^4 - T_0^4) \quad . \tag{7}$$

Rewriting Equation (7) in terms of area, and assuming that the mass of the radiator is generally a temperature-dependant function of area, the mass per unit area of the reference radiator is

$$m_1 = \frac{k(T_1)\ Q_1}{\epsilon\ \sigma\ (T_1^4 - T_0^4)} \quad . \tag{8}$$

A similar expression can be written for the mass per unit area of the heat pump augmented radiator, as in

$$m_2 = \frac{k(T_2)\ Q_T}{\epsilon\ \sigma\ (T_2^4 - T_0^4)} \quad . \tag{9}$$

Now the total mass from the rejection of heat, Q_1, at T_2 is

$$m_T = m_2 + m_r\ Q_1 + m_p\ P \quad . \tag{10}$$

Assuming that k is not a function of temperature, combining Equation (9) in terms of Equation (10), and then dividing by Equation (8) yields an expression for the mass of the heat pump augmented radiator system to that of the reference flat plate radiator, as in

$$m_T/m_1 = \left(\frac{T_1^4 - T_0^4}{T_2^4 - T_0^4}\right) \left[1 + \frac{T_2 - T_1}{e\, T_1} + \frac{(T_2 - T_1)\, T_3}{e\, n\, T_1\, (T_3 - T_2)}\right] + \frac{\epsilon\ \sigma}{k}(T_1^4 - T_0^4) \left\{ m_r + m_p \left[\frac{1 + (T_2 - T_1)}{e\, T_1}\right]\right\} \quad , \tag{11}$$

which is defined as the mass augmentation ratio.

PARAMETRIC ANALYSIS

A mass augmentation ratio of less than unity indicates that the mass of the augmented system is less than that of the flat plate radiator. Because the heat pump augmented radiator system is referenced to a flat plate radiator, parameters that tend to optimize radiator performance will have a detrimental effect on the performance of the heat pump augmented radiator system. Because heat pump augmentation reduces the required radiator area, its advantages will be most pronounced with radiators having a high mass per unit area.

Ten variables are present in the model derived above that, to varying degrees, influence the performance of the heat pump

Table 1 Parametric Variable Groupings

Value Type	Radiator	Heat Pump	Heat Engine
Temperature	T_0, T_1	T_2	T_3
Mass	k	m_r	m_p
Efficiency	ϵ	e	n

augmented radiator system. These variables can be classified into one of three groups: variables associated with the radiator, variables associated with the heat pump, and variables associated with the power system. A tabulation of these variables in their respective classifications is given in Table 1. For this analysis, a nominal value will be assigned to each variable. The presentation will consist of a variation of parameters about several of these values. The operating temperature of a first-generation prime power system will certainly range between 1000 K and 2000 K. This is the high-temperature source of energy that is used to drive a heat engine that supplies shaft power to the compressor of a heat pump. The low temperature energy source is assumed to be related either to an electronics cooling application or to the maintainance of an orbital habitat. The temperature of this heat source would be somewhat higher than room temperature, but not excessively so. For the purposes of this analysis, the source temperature is assumed to be 333 K. The fractional Carnot efficiencies of space power-producing machinery is uncertain. Efficiencies of Earth-bound machinery typically range between 0.01 and 0.6. Efficiencies of 0.3 will be assumed for both the power machnery and the heat pump. Limited information is currently available on the mass of space power systems in the multimegawatt power range. As shown in Dexter and Haskin (1984), systems operating at higher power levels will likely have lower specific masses. The low-end specific mass of high-performance refrigeration machinery is typically 5 kg/kW_t. This will be the nominal value used for the heat pump specific mass. The mass per unit power for the current 100-kW_e SP-100 baseline design is 40 kg/kW_e at 1350 K. Largerscale systems in the megawatt power range will have a lower specific masses than will current systems. A mass per unit power of between 1 and 10 kg/kW would represent an optimistic prediction for the type of systems developed within the next 20 years. For this analysis, a nominal value of 5 kg/kW will be assumed. The mass per unit area of future space heat rejection radiators will certainly range between 1 kg/m^2 and 20 kg/m^2. A mid-range nominal value of 10 kg/m^2 will be used in this analysis.

DISCUSSION

Figure 3 shows the mass augmentation ratio versus sink temperature, with source temperature as a parameter. The improvement of the augmented radiator's performance with increasing sink temperature is due to the relative decrease in the reference flat plate's radiative heat transfer as t e heat source temperature approaches the temperature of space. The improved performances as the source temperature is decreased (the parametric curves) suggest potential for this method in heat rejection systems operating near or below the space sink temperature.

A plot of mass augmentation ratio as a function of source temperature at various heat rejection temperatures is shown in Figure 4. The pocrer performance of the augmented radiator as the heat source temperature increases is due to the higher radiative heat transfer rates of the reference flat plate as its source temperature increases. The increased mass penalty as the temperature of the augmented heat rejection system increases is due to the lower Carnot efficiency of the heat pump as the source temperature increases over the sink temperature. All three curves intersect at the source temperature, corresponding to the minimum possible heat transfer from the flat plate radiator.

The mass augmentation ratio versus heat rejection temperature of the augmented system with prime power source operating temperature as a parameter is shown in Figure 5. Two inflection points are present in the curve generated. The first inflection point is located just above the temperature of the reference flat plate and results from the decrease in heat pump Carnot efficiency. The second inflection point occurs at somewhat higher temperatures and is much more gradual. This inflection is due to lower heat engine efficiencies at elevated heat rejection temperatures. The parameters on this plot show that the locations of the high temperature inflection points increase with prime power system temperature.

A plot of mass augmentation ratio versus prime power operating temperature with heat rejection temperature as a parameter is shown in Figure 6. The performance of the augmented system improves with increasing prime power system temperature. This trend is due to the improved Carnot efficiency of the heat engine as the prime power temperature is increased.

The mass augmentation ratio versus the mass per unit area of both the the reference flat plate and the augmented radiators is shown in Figure 7. The performance of the augmented radiator improves as the mass penalty associated with radiative area is increased. This is due to the difference in temperature between the reference radiator and the augmented one.

Figure 8 shows the relative influence of power system specific mass and heat pump specific mass on the augmentation ratio of the heat pump radiator system. Decreases in power system specific mass will yield larger system mass reductions than will corresponding decreases in heat pump specific mass.

CONCLUSIONS

The results of the following analysis indicate that mass reductions of this one-radiator augmented system over the reference flat plate radiator only occur on the outer range of the parameters varied or near the space sink temperature. Heat pump augmented systems of this configuration do not appear to be promising in the reduction of total radiator mass, given the efficiencies and mass requirements refrigeration machinery currently anticipated. A heat pump augmented sytem of this type might provide mass reductions only in reference radiators, that, for some reason, have an inherently high specific mass. However, in certain instances, when system mass is a secondary concern, such as for electronic cooling purposes or for the enhancement of transportablity by reducing radiator surface area, the use of such systems may be beneficial.

Acknowledgments

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References

Dexter, P. F. and W. L. Haskin (1984) "Analysis of Heat Pump Augmented Systems for Spacecraft Thermal Control," AIAA 19th Thermophysics Conference, June 25-28, Snowmass, Colorado.

Kerrebrock, J. L. (1986) "Optimization of Heat Rejection in Space," AIAA Journal of Propulsion, Vol. 2, No. 6, pp. 562-563.

NOMENCLATURE

A:	Radiator surface area (m^2)
COP:	Heat pump Carnot efficiency [T_2/T_1-1]
e:	Actual heat pump efficiency [ϵ_r COP]
κ:	Radiator mass per unit area (kg/m^2)
m_1:	Mass of reference flat plate radiator (kg)
m_2:	Mass of heat pump augmented radiator (kg)
m_T:	Total mass associated with heat rejection of Q_1 at T_2 (kg)
m_p:	Mass of power system per unit power input (kg/kW_t)
m_r:	Mass of heat pump per unit heat load from T_1 (kg/kW_t)
η:	Actual heat engine efficiency [$\epsilon_e\ \eta_{th}$]
P:	Shaft power (kW)
Q_1:	Load from low temperature source or from reference radiator (kW_t)
Q_2:	Heat rejected by the prime power system (kW_t)
Q_3:	Heat provided by the prime power source (kW_t)
Q_T:	Total heat rejected by heat pump augmented radiator (kW_t)
T_0:	Temperature of space (K)
T_1:	Temperature of reference heat source (K)
T_2:	Temperature of heat rejection system (K)
T_3:	Temperature at exit of prime power system (K)
ϵ:	Emissivity of radiator surface
ϵ_e:	Fractional Carnot efficiency of heat engine
ϵ_r:	Fractional Carnot efficiency of heat pump
η_{th}:	Heat engine Carnot efficiency [$1-T_2/T_3$]
σ:	Stefan-Boltzmann constant (5.67×10^{-5} $kW/m^2\ K^4$)

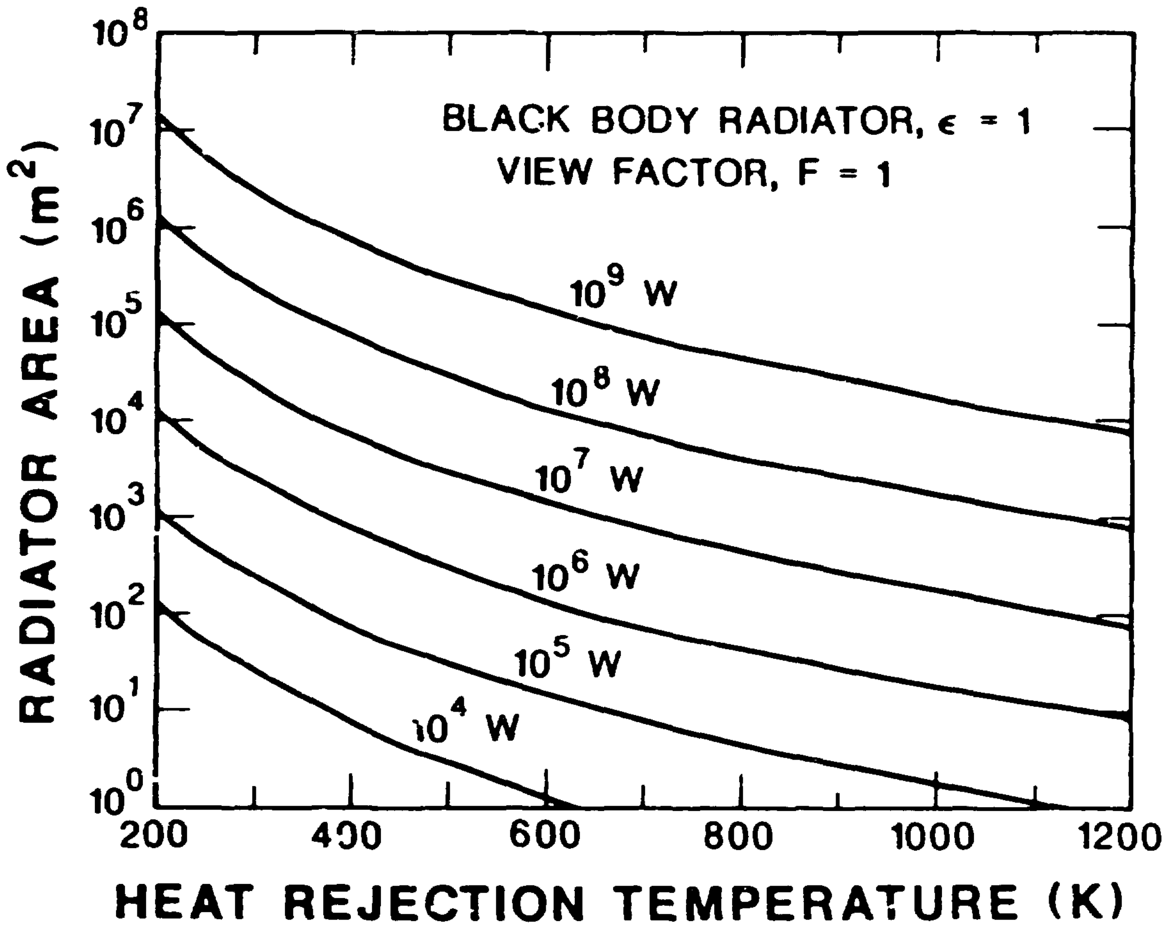

Figure 1 Surface Area Versus Heat Rejection Temperature with Heat Load as a Parameter for an Idealized Flat-plate Black Body Radiator to a 0 K Heat Sink.

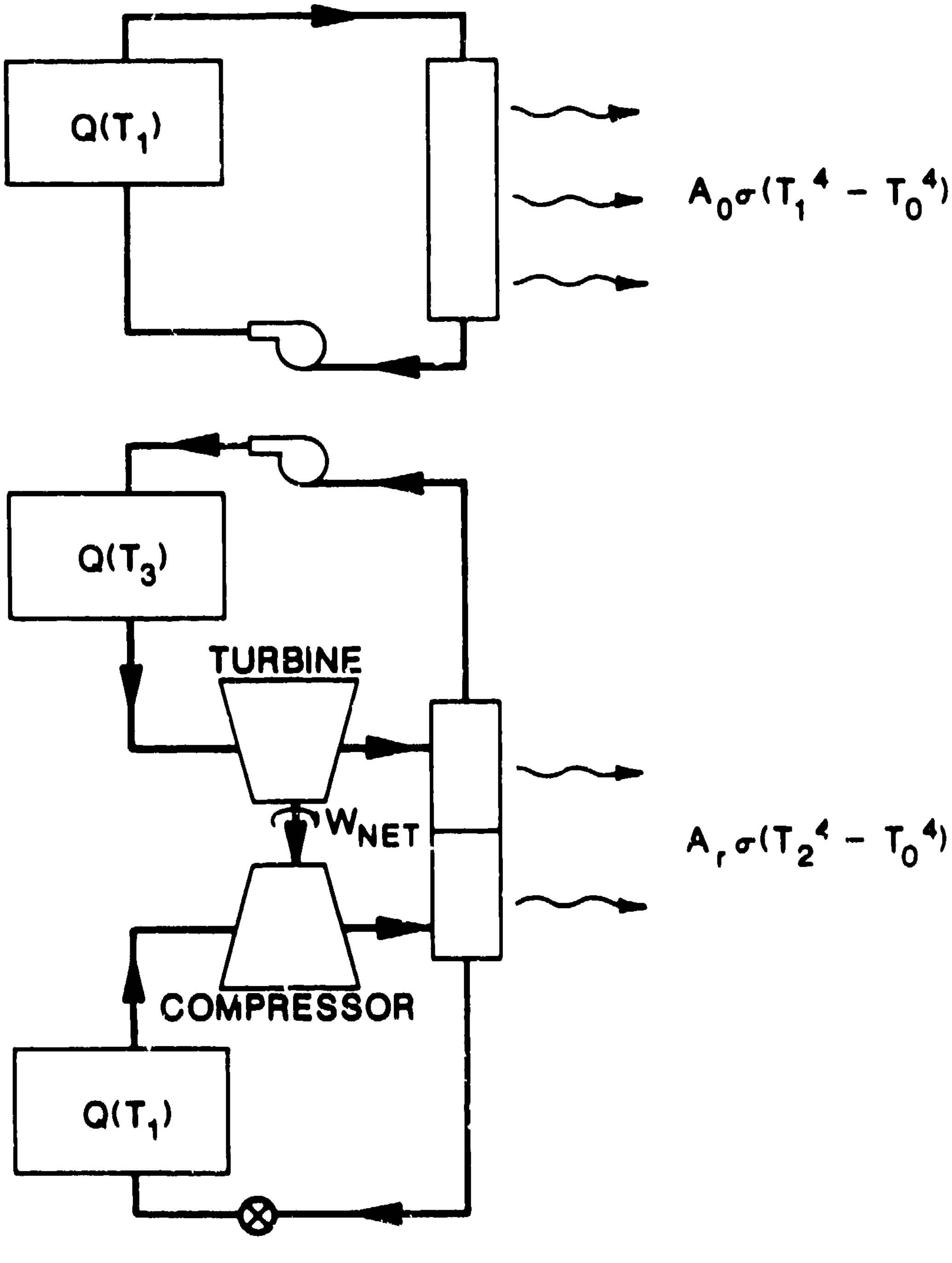

Figure 2 Schematic Diagram of the Proposed Heat Pump Augmented System.

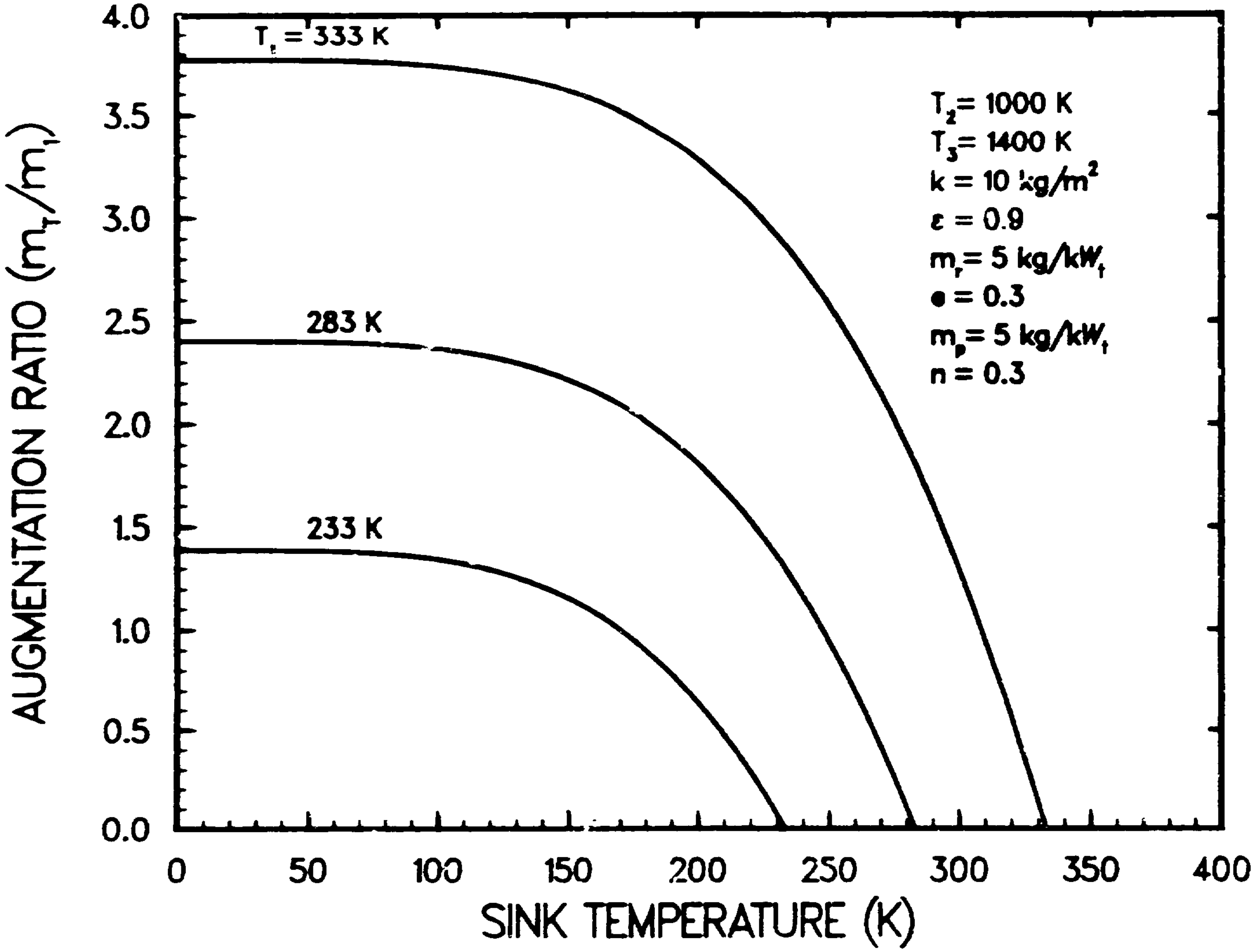

Figure 3 Mass Augmentation Ratio Versus Sink Temperature with Heat Source Temperature as a Parameter.

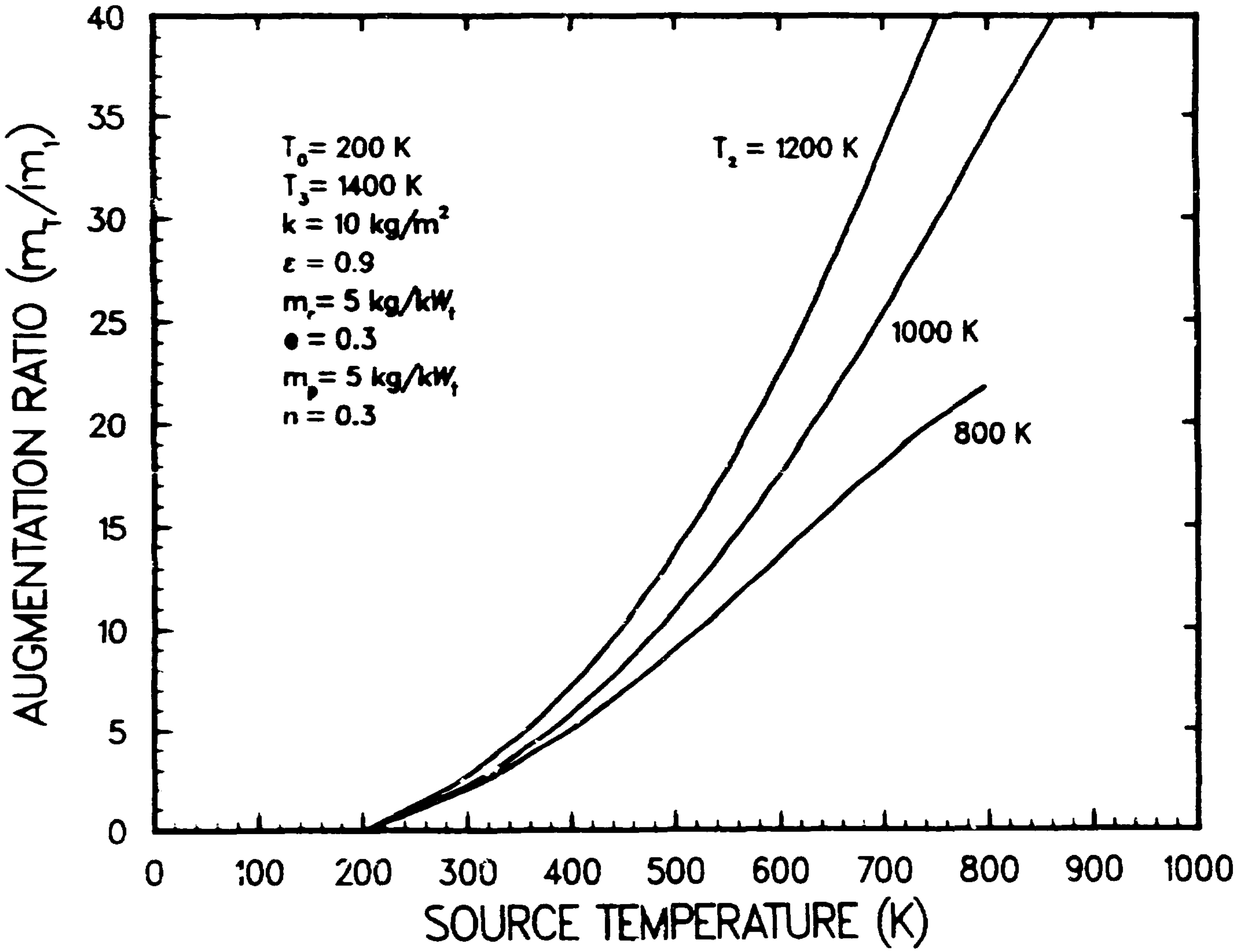

Figure 4 Mass Augmentation Ratio Versus Source Temperature with Heat Rejection Temperature as a Parameter.

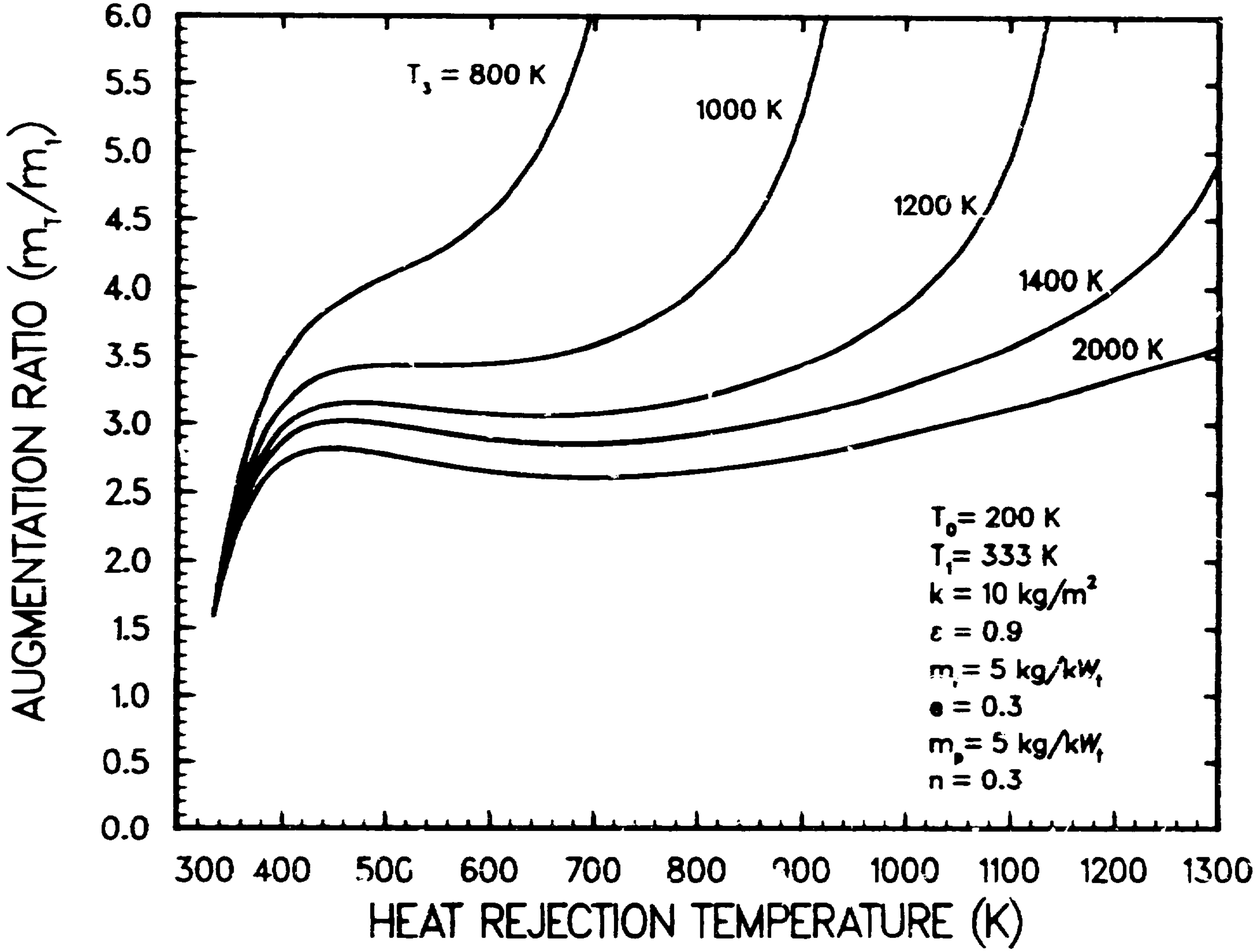

Figure 5 Mass Augmentation Ratio Versus Heat Rejection Temperature of the Augmented System with Prime Power Source Temperature as a Parameter.

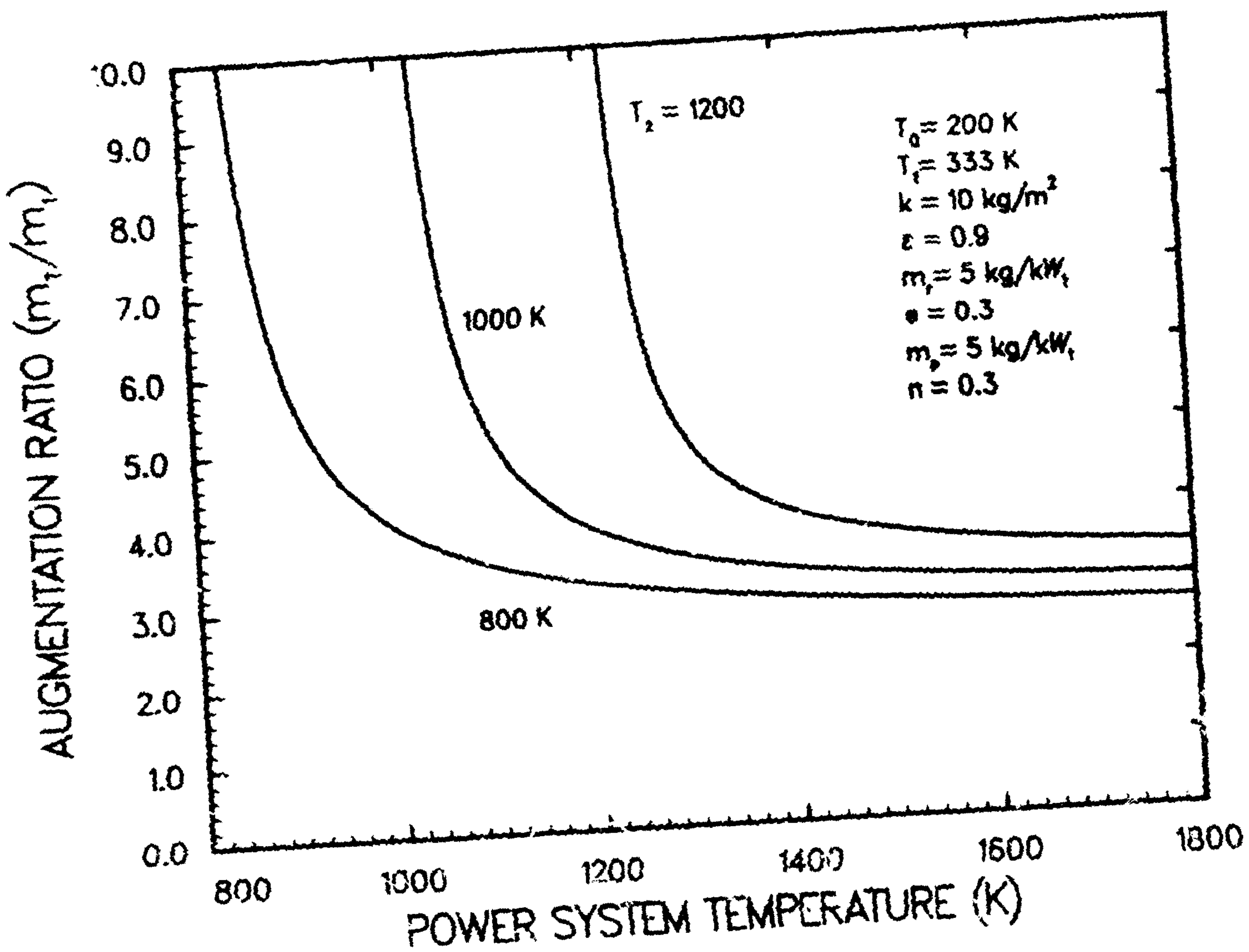

Figure 6 Mass Augmentation Ratio versus Prime Power Source Temperature with Heat Rejection Temperature as a Parameter.

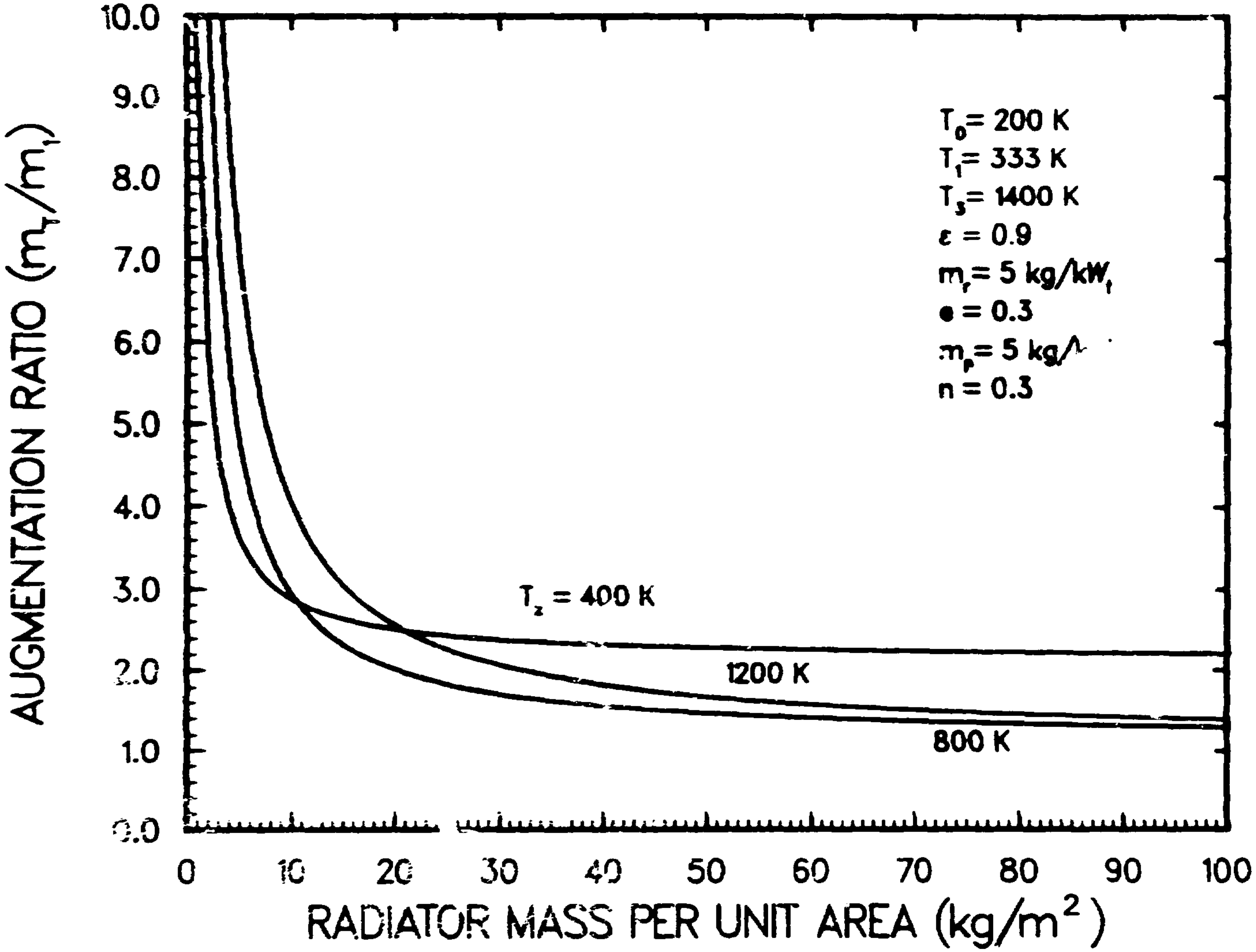

Figure 7 Mass Augmentation Ratio Versus Radiator Mass Per Unit Area Ratio with Heat Rejection Temperature as a Parameter.

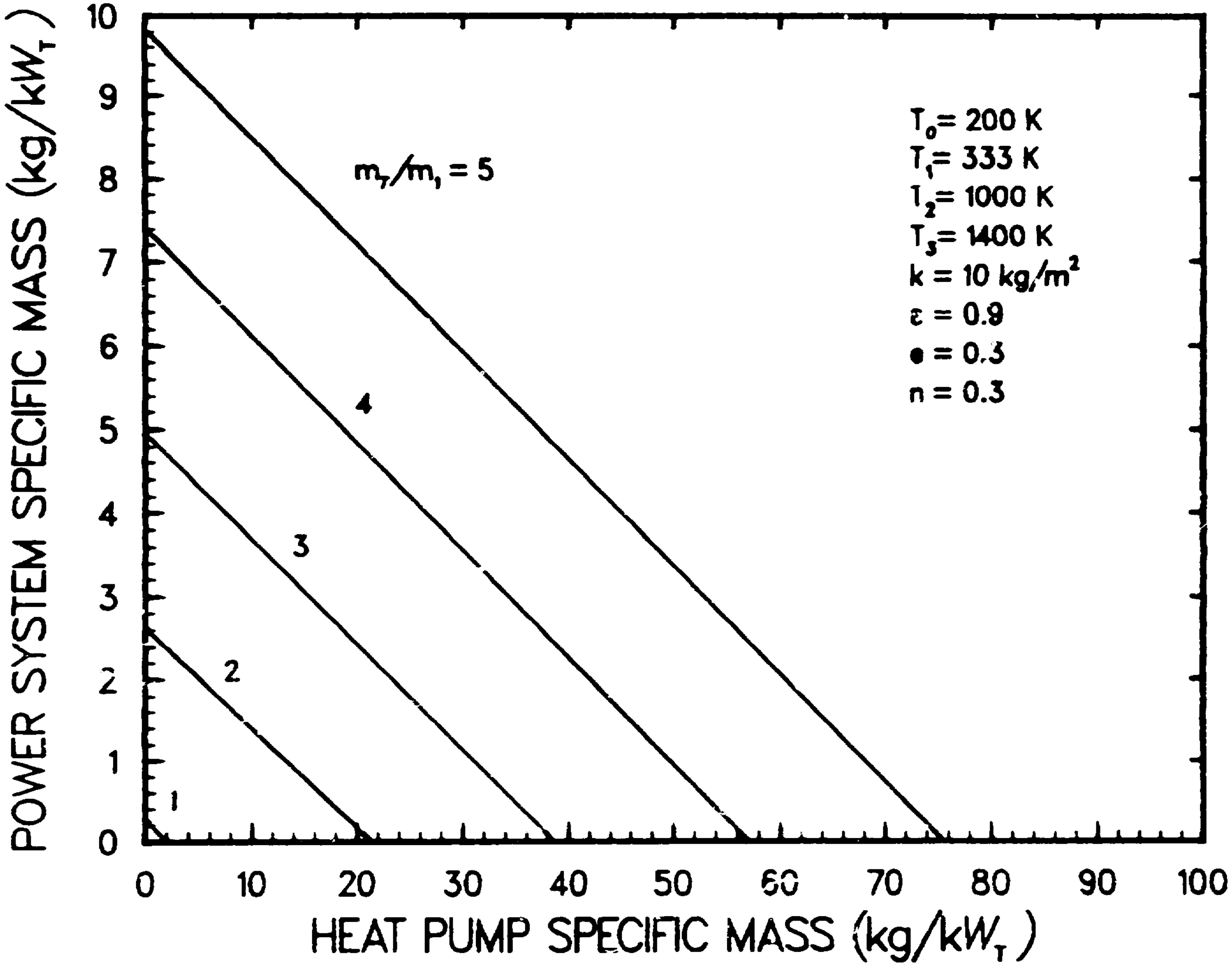

Figure 8 Power System Specific Mass versus Heat Pump Specific Mass with Mass Augmentation Ratio as a Parameter.